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TWO-PHASE/TWO-PHASE HEAT EXCHANGER ANALYSIS

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TWO-PHASE/TWO-PHASE HEAT EXCHANGER SIMULATION ANALYSIS

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SUMMARY

A capillary pumped loop (CPL) system with a condenser linked to a double two-phase heat exchanger is analyzed numerically to simulate the performance of the system from different starting conditions to a steady state condition based on a simplified model. Results of the investigation are compared with those of similar apparatus available in the Space Station applications of the CPL system with a double two-phase heat exchanger.

INTRODUCTION

A capillary pumped loop (CPL) system consists of an evaporator pump, a condenser, a subcooler, a vapor transport line, a liquid transport line and a storage tank with a possible starter pump (Ku and others, 1986; Neiswanger and others, 1987; Kim, 1990). The condenser structured as a concentric core of a double tube heat exchanger dissipates heat energy to an evaporator, which is the annulus section of the double tube exchanger, so that the fluid in the core condenses while the fluid in the annulus section evaporates. A CPL system equipped with a condenser having a two-phase/two-phase double tube heat dissipation structure (hereafter, referred to as "DBLHX") is to be simulated numerically. A schematic block diagram is shown in Fig. 1.

The system has two different types of evaporator pumps, each of which has twelve units of evaporator pumps, a DBLHX system, which has six DBLHX units as shown in Fig. 2. The saturated liquid coming out of the TEMP2C-condenser enters a subcooler next to the DBLHX system with a non-condensable gas collector (Fig. 1), for further reduction of energy from the liquid then passing through a long liquid transportation line. The subcooled liquid enters the system evaporators; one is a cold plate type and the other is of a hybrid type. The system evaporator consists of several units of a circular tube heat exchanger in parallel, all of which are attached to a plate. Each circular tube exchanger has an isolator so that each circular tube exchanger performs independently of the other in case of individual unit failures. The tube exchangers have internal grooves which have a porous outer layer. As the subcooled liquid enters the core section of the tube, after having passed the isolator, the fluid permeates the porous layer and evaporates at the interface of the groove and the porous layer as a result of the heat transferred from the attached plate, which is then pushed into the grooves, entering the vapor header which is common for the circular tube exchanger. The vapor passes through the vapor line

and enters the DBLHX. By doing so, the CPL completes its cycle of transferring heat from the system evaporator to a sink. The sink would be a chiller system of a refrigeration system or surroundings of an orbiting satellite and the Space Station.

A chiller system with refrigerant R114 removes the heat energy from the DBLHX and subcooler in a CPL system which has been in operation for some time in the NASA, GSFC Greenbelt, MD. Limited experimental results are available.

A schematic block diagram of the CPL system shown in Fig.1 reveals a sub-system of control devices of its flow. Fig.1 exhibits two different types of evaporator pumps, each of which has twelve units of evaporator pumps (EVPs), six units of DBLHX as shown in Fig.2, the condenser of the DBLHX, which is designated TEMP2C-condenser, the evaporator of the DBLHX, referred to as the TEMP3C-evaporator, the TEMP2C-subcooler with a noncondensable gas collector (Fig.1), long liquid and vapor lines, a subsystem of control devices consisting of valves, fluid meters, and a reservoir with a starter pump. Fig.2 shows the plan and elevation views of the TEMP2C/TEMP3C DBLHX. Flow arrangement reveals that ammonia flows in three different levels of elevation. Important dimensions of this subsystem are shown in Fig.2c. A CPL system without a buffered evaporator has been successfully modeled (Cullimore, 1990, Neiswanger et al., 1987).

The CPL system with the DBLHX has not yet been investigated. If the internal tubing does not have the internal and external grooves and if phase changes in the media do not take place, the device is a double-tube heat exchanger for which analysis has well been established in literature (Eckert, 1962; Holman, 1990).

In order to have a manageable simulation system, certain assumptions are made: (1) A cold plate evaporator pump replaces a hybrid evaporator so that two identical cold plate evaporator pumps are a set of evaporator pumps; (2) Fan-shape inlet and exit sections of the DBLHX (Fig.2) are assumed to be an assembly of six straight tubes in the same level of elevation; (3) The non-condensable gas (NCG) collector in the subcooler does not affect the performance of the system; (4) The flow control devices for the system and the capillary starter pump are not included in the model in question; and (5) The storage tank behaves as an infinite source with constant properties; (6) As a first approximation, the connecting tubes between the components are not important and consequently are not included in the system modeling except for the long ammonia transportation lines of vapor and liquid; and (7) The porous material has no thermal resistance with the fluid at the same temperature. Such assumptions result in the block diagram as shown in Fig.3.

Starting from the end of the subcooler, a few key points are identified as J1001, J1002, ..., J1005 along with the Line 101, PLEN 999, evaporator, Line 401, TEMP2C condenser, TEMP3C evaporator, TEMP2C subcooler and R114 chiller. A code known as SINDA'85/FLUINT (Cullimore and others, 1989) is employed for the simulation scheme and notations to those of SINDA'85/FLUINT are used in Fig.3. Details of these components consisting of an entire

simulation model are illustrated in the following sections.

EVAPORATOR PUMP SYSTEM

Cross-sections of the evaporator pump in two directions are shown in Figs.4a and 4b. Fig.4c illustrates how the liquid flow evaporates. The cold plate evaporator pump(EVP) with twelve(12) EVP's,twelve isolators are shown in Fig.4d. Fig.4e explains the cross-section of the circular evaporator pump(EVP), and the attached plate and isogrid structure are illustrated in Fig.4e. Based on this figure of the combined cross-section, an approximate method for the extended surface theory is used. Namely, at the mid point of the plate, the temperature is the maximum and the temperature at the tube is a fixed temperature. Therefore, an approximate fin efficiency is used for an average temperature(or conductance) as far as the plate is concerned(see Fig.4e for an illustration). The radial cross-section of the EVP has 30 internal grooves in the outer shell. From a header, a liquid enters the isolator(Fig.4g),permeates the porous layer, reaching the core of the EVP. Once the liquid reaches the groove surface through permeation,heat transfer from the outside causes vaporization of the liquid. Evaporated fluid is pushed to the grooves(Fig.4c) and to the vapor header and enters the vapor transport line. The capacity of the EVP system is assumed to be 3600 w.

DOUBLE TWO-PHASE HEAT-EXCHANGER SYSTEM

In the DBLHX, the condenser is a inner circular tube coupled with an evaporator which is the annulus section of the DBLHX. The inner tube has axial grooves internally and externally and porous material layers occupy the space next to the grooves in the annular section and the core section, respectively,so that liquid,from capillary action can permeate the grooves and the porous layers(Fig.5). Six DBLHXs connected in parallel function as the condenser. The layer of porous material,Porex,enhances condensation in the internal grooves. Six DBLHX evaporators connected in parallel are the TEMP3C-evaporator. The layer of Porex directs the liquid flow in one desired direction. Porex has a permeability of $2.60E-13$.

From the inlet of TEMP2C-condenser,a vapor enters internal grooves of the internal tube,condenses on the surface of the porous layer and permeates the porous layer,reaches the core cavity and leaves the TEMP2C-condenser from the core section of the DBLHX. From the inlet of TEMP3C-evaporator(the exit side of TEMP2C-condenser),liquid flows into the annulus section where the porous layer is placed. The porous layer has grooves in the metal outer shell side. As the liquid permeates the porous layer,it reaches the external grooves of the internal tube,starts evaporating because of heat transferred from the condenser, then is pushed into the grooves, leaving TEMP3C-evaporator from the annulus section. From

this exit, it is possible for a mixture of saturated liquid and vapor to leave TEMP3C-evaporator.

Figs. 5a and 5b show an axial cross-section and a radial cross-section, respectively. Six of the fan-shaped inlet and exit sections are represented by a single straight tube of the same size at the same elevation then duplicates six times. The core section from which the liquid leaves TEMP2C-condenser has an isolator then a liquid header follows. Due to heat transfer from TEMP2C to TEMP3C across the metal tube, it is plausible to consider that the quality of condenser fluid changes from unity to zero, while the quality of evaporator fluid changes from zero to a number which is close to unity, say 0.8. For simplicity, the variations of fluid quality in TEMP2C and TEMP3C are assumed to be zero in the beginning half length of the DBLHX and unity in the rest of the length of the TEMP2C-condenser, but 0.8 in the rest of the length of the TEMP3C-evaporator of the DBLHX. This is a case where the DBLHX is divided into two sections, and the two sections have two different qualities initially. If the entire length of the DBLHX is divided into N equal number of segments, the result of computation can be improved. The number N may be 10 or 20, or 30. Accuracy of solutions corresponding to the number of segments will be discussed later.

SUBCOOLER AND R114 CHILLER

The subcooler of the CPL has a complicated structure because of a vapor trap necessary to cope with non-condensable gas in the subcooler. Considering that the amount of noncondensable gas is relatively small in comparison with the flow rate, the vapor trap is not modeled in the present study. In the location of the trap, a flat plate is placed (Fig. 6). Similarly to the evaporator pump plate, the half of the plate with the attached subcooler and the R114 chiller is considered to be like a fin having the minimum temperature at the mid point of the contacting area between the subcooler and the chiller (Fig. 6). The fin efficiency is assumed to be 80 per cent. The temperature variation along the transverse direction is assumed to be rather small in comparison to that in the direction of the tube axis. For each leg of the subcooler and the chiller a uniform temperature is assumed to exist and the 180 degree bends connecting four legs are considered to be adiabatic.

The subcooled liquid passes through the liquid transportation line which is considered adiabatic. A pressure drop through this passage is added to the system pressure loss. The liquid then enters the evaporator pump. The liquid reservoir is added to the system model as a plenum, holding all the properties as constant. A mixing process in the reservoir may not result in constant properties, nonetheless, it is assumed a steady state process.

INPUT FILE FOR THE COMPUTER PROGRAM

In the simulation here, a steady state simulation is sought under different initial operation conditions. SINDA'85/FLUINT has

subroutines for steady state solutions for thermal and fluid nodes. SINDA'85/FLUINT is a structured code employing some of FORTRAN commands and the input/output of data for thermal, fluid nodes and operation execution follow the certain format as follows:

1. Header Options Data
2. Header Node Data, PLATE
3. Header Conductor Data, PLATE
4. Header Control Data
5. Header Source Data, PLATE
6. Header Flow Data, TEMP2C
7. Header Flow Data, TEMP3C
8. Header Flow Data, STSL
9. Header Flogic 0, TEMP2C
10. Header Variables 1, PLATE
11. Header Output Calls, TEMP2C
12. Header Operation Data
 - Build DBLHX, PLATE
 - Buildf DBLHX, TEMP2C, TEMP3C, STSL
 - Call FASTIC
 - Call STDSTL
13. End of data

The term PLATE is the name of the thermal node system which consists of the EVP tube, its plate, isogrid structure, internal and external tubes of the DBLHX units, the subcooler tubes, its plates and tubes of R114 subsystem. TEMP2C represent the fluid submodel for the EVPs, TEMP2C-condenser, and the subcooler. TEMP3C is made of the buffered evaporator of DBLHXs. STSL represents the fluid subsystem which is a loop with refrigerant R114. Execution of the program starts with consolidating data on PLATE, TEMP2C, TEMP3C, and STSL by the commands Build and Buildf and computes the energy balance among the nodes and fluid submodels by calling FASTIC and STDSTL in the Header Operation data. These subroutines compute the thermal network in steady state. FASTIC approximates the fluid conduit as frictionless passages and the fluid lump as if it does not have inertia. After this routine, as STDSTL is applied to the result of FASTIC, a refined solution for temperatures is obtained. The basic flow in FASTIC and STDSTL is illustrated in Fig.7. The terms used in the flow chart are explained in the NOMENCLATURE section. Essentially, solutions obtained by FASTIC and STDSTL are the roots of the equation $Q_i = f(T_i^{k+1}, T_i^k, G)$ where Q_i is the heat rates, T_i^{k+1} , T_i^k represent the i-temperature of nodes at iteration numbers of k+1 and k-th, respectively, and G is the conductance of the node and its neighborhood nodes.

Details of these headers are explained in the manual (Cullimore and others, 1989). Node and Conductor Data sections have the initial temperature, capacitance and conductance of these submodels. Transportation lines for liquid and vapor are adiabatic. The numbers for nodes, conductors, lumps and connections of the EVP's are in 200's and the remaining numbers used for the thermal and fluid submodels are listed in Table 1.

Table 1 Diffusion and Fluid Model Identifications
by a Range of Numbers

Liquid Transportation Line	100 to 199
Evaporator	200 to 399
Vapor Transportation Line	400 to 450
TEMP2C-Condenser	500 to 599
TEMP2C-Subcooler/R114 Chiller	600 to 699
TEMP3C-Evaporator	700 to 799
R114 Chiller	800 to 899
Storage Tank	900 to 999

Table 2 summarizes the proper dimensions of subsystems and flow characteristics.

Table 2
Proper Dimensions of Evaporator, DBLHX and Subcooler

Evaporator:	
Flow rate.....	0.001 kg/s
Number of Cold Plates.....	2
Number of Evaporator pumps.....	12
Effective Length of Evaporator Pump.....	0.648 m
OD of Evaporator Pump.....	0.0158 m
ID of Evaporator Pump.....	0.0112 m
Number of Internal Grooves.....	30
Depth of the Grooves.....	0.0010 m
Width of the Grooves.....	0.0012 m
Outer Radius of the Porous Layer.....	0.0106 m
Inner Radius of the Porous Layer.....	0.0090 m
Plate Width of the Evaporator Pump.....	0.1108 m
Plate Thickness.....	0.0035 m
Isogrid Structure thickness.....	0.0025 m
Isolator Length.....	0.0254 m
Isolator Outside Diameter.....	0.0190 m
Isolator Inside diameter.....	0.0172 m
Porex Inside diameter.....	0.0003 m
DBLHX TEMP2C-Condenser:	
Effective length of DBLHX.....	0.6880 m
Number of TEMP2C-Condenser Units.....	6
Number of Internal Grooves.....	45
Depth of the Grooves.....	0.0010 m
Width of the Grooves.....	0.0012 m
Outer Radius of the Porous layer.....	0.0222 m
Inner Radius of the Porous Layer.....	0.0181 m
Isolator Length.....	0.0381 m
Isolator Outside Diameter.....	0.0111 m
Isolator Inside Diameter.....	0.0104 m
Porous Layer Inside Diameter.....	0.0060 m
DBLHX TEMP3C-Evaporator:	
Number of External Grooves.....	45
OD of TEMP3C-Evaporator.....	0.0428 m

ID of TEMP3C-Evaporator.....0.0381 m
 Depth of the Grooves..... 0.0010 m
 Width of the grooves..... 0.0012 m
 Outer Radius of the Porous Layer.....0.0381 m
 Inner Radius of the Porous Layer.....0.0307 m
 Subcooler:
 Length of the Leg..... 1.03 m
 Number of Legs..... 4
 Outside Diameter of tube..... 0.0119 m
 Inside Diameter of tube.....0.0104 m
 Attached Plate Length.....0.6688 m
 Plate Thickness.....0.0050 m
 R114 chiller:
 Number of Legs.....4
 Outside Diameter.....0.0199 m
 Inside Diameter..... 0.0157 m
 Flow Rate.....0.35 kg/s
 Liquid Transportation Line:
 Length.....25 m
 Number of Turns.....18
 Outside Diameter.....0.0146 m
 Inside Diameter.....0.0104 m
 Vapor Transportation Line:
 Number of Turns.....18
 Outside Diameter.....0.0203 m
 Inside Diameter0.0189 m

N-segments are used in modeling the TEMP2C-condenser. The first device of the N-segments is a tank and from then on, N lumps representing N segments transfer energy through N ties to the neighboring nodes, and the liquid passes through the wick from capillary action(CAPIL). Initial qualities of the fluid are assumed according to the locations of the segments linearly. The quality of the TEMP2C-condenser varies from unity to zero. The same technique is applied to the N-segment TEMP3C-evaporator with initial quality variations of zero extending to 0.8.

To the TEMP2C-subcooler coupled with the R114 chiller system, a macro command HX(Cullimore,1990) was applied with four sections. Due to the added plate to the two mid sections of this subsystem, the macro command may not produce an exact simulation result. The fluid transportation lines are modeled by a macro command LINE.

RESULTS

Ammonia is used as a medium. Except for the isolator tubing and fittings for plumbing which are stainless steel, tubing and plate used for the CPL system are aluminum. The thermal properties of these materials used for computation are from the room condition. They are listed in Table 3.

Table 3
Thermal and Other Properties of Materials

	aluminum	stainless steel
Density	2659 kg/m ³	8266 kg/m ³
Specific Heat	867 J/kgK	46 J/kgk
Thermal Conductivity	164 W/m K	12.8 W/m K

The properties of the porous material, Porex is not needed because an assumption was made in such a way that these do not affect the flow condition excluding the capillary action in the porous layer. The permeability of Porex is 2.6E-13.

One of the objectives of the simulation analysis is to obtain a steady state solution from the simulated model. The first step of a well defined computation scheme is to establish the number of iterations needed for reaching a steady state condition. The available schemes for the steady state computation are FASTIC, and STDSTL (Cullimore, 1990). FASTIC should be employed first and then applied is STDSTL. By doing so, a steady state solution is obtained. In order to determine the iteration number, LOOPCT for FASTIC and STDSTL, the entire system is placed at a surrounding temperature of 30 degree C while the chiller system is at 8.0 degree C. Two sections of the DBLHX and 200 iterations (LOOPCT=200) were tried with the input file. Some of the model simulation were macro commands such as CAPPMP, LINE, and HX (Cullimore, 1990, CPL example) and were applied to EVP, transportation lines and the subcooler, respectively. It is seen from the example by Cullimore that the use of macro commands is reasonable, therefore, they are used, here. The EVP is considered as one unit maintaining a fixed evaporating temperature, the transportation lines area adiabatic as one unit each, and the subcooler is considered as four sections with the 180 degree connecting tubes being adiabatic. Computation was run with 200, 500, and 1500 iterations under FASTIC and STDSTL. It was discovered that with iterations of 500, a STDSTL computation was found reasonable, even though the energy balance set by the program was not met. However, the energy into and out of the system is within 0.3 percent. The result of computation with iterations of 1500 did not yield a significant difference from those from 500 iteration computations. So, it was determined that further computation was done with 500 iterations.

The number of divisions on the DBLHX was made into 2, 10, and 20 and the result of energy into and out of system is listed in Table 4.

Table 4
Solution Convergence with Number of Divisions of DBLHX

	No. of Divisions		
	2	10	20
FASTIC Energy in	2526.72 w	2536.25	2533.17

	out 2518.95	2536.91	2532.47
STDSTL Energy in	2483.61	2528.23	2533.17
	out 2469.04	2509.95	2532.47

As seen from Table 4, the less divisions of the DBLHX, the larger the discrepancies are between the energy in and out of the system and between the results of FASTIC and STDSTL. Computation of different surrounding temperatures of 8.0, 30.0, and 50.0 degrees C (hereafter referred to as Cases I, II, and III), therefore, was carried out with 20 divisions of the DBLHX and 500 iterations.

Table 5 lists the capacity of the system which starts from the surrounding temperatures of the three cases. The capacities from the different surrounding temperatures seem to converge to a range of 2428.00 to 2533 w.

Table 5
Capacities of the System Approaching from Different Initial Conditions

	Case I	Case II	Case III
STDSTL Energy in,w	2428.39	2533.17	2526.56
Energy out,w	2437.04	2532.47	2526.60

In Case I, the discrepancy between the energy in and out of the system is the largest, but in the other two Cases II, and III, the discrepancies are negligible. The efficiency of the system at the design condition is about 70.00 per cent. Table 6 presents the temperature distributions of PLATE, i.e., the evaporator, evaporator plate, TEMP2C-Condenser, subcooler for the three cases. It is seen from Table 6 that the temperature at node 519 is 13.92 of Case I and the temperatures at node 501 and 519 of Case III is 22.18 and 23.69 showing sudden changes in temperature variations. The sudden change in temperature will be explained in the DISCUSSION section. The temperature reaches about 30.0 degree C in the evaporator, the TEMP2C-condenser and reduces down to 21.99, 24.28, and 24.17 degrees C, respectively, in the subcooler for the three cases.

Table 6
Temperature Distributions of PLATE, Evaporator, TEMP2C-Condenser and Subcooler

Node Id.	Case I	Case II	Case III
200	31.38	30.82	31.34
201	31.38	30.82	31.34
501	22.18	30.10	29.40

503	30.15	30.10	30.06
507	30.10	30.12	30.15
511	29.74	30.15	30.16
515	29.59	30.16	30.15
519	13.92	30.10	23.69
520	22.85	30.10	30.04
651	22.51	22.61	23.86
652	23.67	24.17	24.47
653	23.68	25.34	25.32
654	21.99	24.28	24.17

Table 7 shows the temperature and quality distributions for the fluid starting from the liquid line to the subcooler along the length of the conduit for the three cases.

Table 7
Temperature and Quality Distributions along the Fluid Conduit

Lump Id.	Case I		Case II		Case III	
	T, C	X	T, C	X	T, C	X
1001	30.09	0.1818	30.15	0.3898	30.15	0.3732
101	30.05	0.1822	30.09	0.3902	30.09	0.3734
140	30.03	0.1822	30.03	0.3904	30.06	0.3735
1002	30.03	0.1968	30.00	0.3905	30.05	0.3856
210	30.03	0.1808	30.05	0.3796	30.05	0.3665
220	29.93	0.0000	30.01	0.0000	29.99	0.0000
230	29.19	1.0000	30.25	1.0000	30.20	1.0000
520	30.15	1.0000	30.21	1.0000	30.21	1.0000
523	30.10	0.9640	30.17	0.9580	30.17	0.7031
527	30.10	0.9647	30.17	0.9172	30.16	0.6714
533	30.10	0.9621	30.17	0.9019	30.16	0.6442
537	30.10	0.9620	30.17	0.8985	30.16	0.6315
540	30.09	0.7187	30.16	0.9992	30.16	0.9781
1005	30.09	0.7187	30.16	0.9800	30.16	0.9781
601	30.09	0.5787	30.16	0.7800	30.16	0.7613
602	30.09	0.3878	30.16	0.6211	30.15	0.6035
603	30.09	0.2832	30.16	0.5064	30.15	0.4892
604	30.09	0.1818	30.15	0.3896	30.15	0.3732
999	30.00	0.0000	30.01	0.0000	30.00	0.0000

In order to visualize the characteristic behavior of the quality of the TEMP2C-condenser and subcooler at a glance, the quality of the lump locations above is plotted in Fig.8. The horizontal axis represents the relative node locations with respect to the neighboring nodes, and does not indicate the distance between the nodes. It is seen from Table 7 and Fig.8 that the quality is

continuously decreasing in Case I but Cases II and III have a hump around node 540 which is the end of the TEMP2C-condenser. At the end of the TEMP2C-condenser, the medium exists in a higher quality saturation condition than the some of the mid-sections. The quality drops down rapidly in the subcooler section, but remaining as a saturated liquid. The end condition of the TEMP2C-condenser for three cases is not what is sought for at all.

A summary of pressure drops from a few of selected paths of the CPL system is shown in Table 8. There are paths with negative numbers in the pressure drops, implying reverse flows existing in the system. The first negative pressure drop occurs at the end of the evaporator in all of three cases. Then there it occurs around the entrance and exit of the evaporator and the entrance of the TEMP2C-condenser where it coincides with the positions of the quality humps.

Table 8
Pressure Drops in Pa from Paths for Three Cases

Path Id.	Case I	Case II	Case III
101	1262	2048	2003
102	608.4	1005	820.7
140	255.8	578.8	365.9
111	- 2.801	-.1985	-9.416
210	-506.1	-2164.0	-1849.0
260	-1850.0	-4.226E4	-1902.0
403	33.11	31.75	36.03
421	-1.3566E-4	129.4	123.3
422	135.5	80.07	91.75
440	174.2	240.3	229.0
501	1203.0	1024.0	1092.0
520	-4.131	-1.895	-9.716
521	14.53	22.17	22.06
530	14.59	21.96	7.098
540	6.190	1.550	3.593
506	273.4	371.3	300.4
601	35.01	51.57	40.79
602	79.85	99.75	108.1
603	63.16	91.69	92.52
604	24.02	41.62	39.22
605	901.9	1603.0	1650.0

The temperature and quality variations and pressure drops of the TEMP3C subsystem for Case II are attached to the Appendix.

DISCUSSIONS

Discussions are presented on items in the order of appearance in the RESULTS section.

1. The three initial surrounding temperatures were used to achieve a steady state condition by way of FASTIC and STDSTL. Table 3 shows that the larger the number of segments for the DBLHX is, the closer the capacity of the system becomes between the results. The results of twenty divisions of DBLHX seem to indicate good trade-off results between the accuracy and the CPU time. Table 4 shows that the results of FASTIC computations are different from those of STDSTL. The manual recommends the results of STDSTL computations as the final solutions. Case I yielded a low capacity for the system, but it is in the same order of magnitude of the capacity for the system as those of Cases II and III. The efficiency of the system is about 70 percent.

2. The temperatures from the evaporator through the TEMP2C-condenser and the subcooler of the system PLATE are about at 30.0 degree and they decrease in the subcooler sections in three cases. The lowest temperature was at the end of the subcooler for Case I (see Table 6). Table 7 explains a reason for this behavior.

3. Table 7 shows that the temperature and quality are listed side by side at selected lumps from the evaporator to the subcooler. The quality starts with zero from the evaporator, changes to unity remains at unity through the TEMP2C-condenser and reduces toward zero in the subcooler for the three cases. Initial assigned qualities are from unity to zero for the TEMP2C-condenser and remained zero in the subcooler region. The general tendency of the quality distribution is to start with unity, decrease to zero. At the exit of the TEMP2C-condenser, the quality of ammonia does not reach that of the liquid and even after the subcooler, the vapor still remains in the conduit and the saturated ammonia is carried to the liquid transportation line. This occurred in all of the three cases. A probable reason for this would be that the liquid passage of each section of the TEMP2C-condenser is not provided individually in modeling the TEMP2C-condenser. A second reason may be an inappropriate condensing heat transfer coefficient provided by the built-in subroutine which was obtained from the fully developed bubbly flow in a circular tube (Chen, 1963; Traviss, 1973). In the TEMP2C-condenser, condensation takes place in the interface between the grooves and the porous layer. This is not a fully developed flow, as a result, the coefficient should have been much greater.

Considerable amount of energy reduction from the TEMP2C-condenser should have been realized, so that the quality should have reached zero at the end of the TEMP2C-condenser. Further temperature drop would have been made in the subcooler. Instead, only in the subcooler, a continuous temperature drop is demonstrated.

4. As seen in Table 8, there are a few node locations where "negative" pressure drop occurs, implying reverse flows in those locations. However, the locations seem to be places where area reduction takes place in the conduit so that an acceleration exists or a discontinuity or irregularity of quality distribution takes place as these node locations are investigated against those humps

of quality in Table 7. The local acceleration due to the area change may cause a pressure drop, consequently, the overall pictures of the pressure drop and the quality distribution of the fluid passage will be upset from the current modelling cases. This means that a refinement of modeling of these areas is needed by including a local acceleration with a finer section.

5. The comparison with experimental values of temperatures and pressures is made in Table 9. In order to compare the experimental results to the simulated result, a set of pressure and temperature of the evaporator and the TEMP3C-evaporator was selected from an experimental data set and these were used as inputs to the file constructed explained in the above.

Table 9
Comparison of Experiments with Simulated Results

			Experiments	Simulation Results
Pr.-Evaporator,	pa		1.5776E6	1.0124E6
Pr.-TEMP3C-Evp.,	pa		1.4530E6	1.0024E6
Power, in,	w		1792.6	3600
out,			1520.3	2223
Efficiency	%		84.8	61.8
Flow Rate,	kg/s		0.00146	0.001
TEMP2C,	T _{in} ,	C	27.5	25.32
	T _{out} ,	C	26.9	25.25
Subcooler,	T _{in} ,	C	18.9	25.25
	T _{out} ,	C	13.0	25.24
TEMP3C,	T _{in} ,	C	24.1	25.27
	T _{out} ,	C	16.7	25.26

The difference between the evaporator pressures of the experiments and simulated results is significant. The order of the magnitude of the difference between the TEMP3C-evaporator pressures is the same as that of the evaporator pressures. The flow rate of the experiments was calculated to be 0.00146 kg/s, while the simulated result used 0.001 kg/s for the flow rate. This difference would raise the pressure in the TEMP2C and the heat transfer.

Results of experiments show that a slight temperature difference exists in the TEMP2C-condenser and a reduction of temperature in the subcooler implies that the subcooler really removes energy from the saturated liquid, while the simulated results demonstrate almost a constant temperature in the TEMP2C-condenser where condensation takes place. In the subcooler at a lower temperature than the TEMP2C-condenser temperature, condensation takes place in a saturation condition. This is seen from Table 7 and Fig.8.

However, the simulated result and the experimental result both show the similar trends in their reduction in temperature in the

TEMP2C and the subcooler, even though temperature reduction in the simulated result is less than in experiments. As pointed out, modelling will need further refinements in the area of phase change phenomena. For the sudden cross-section area change in general, more detailed, closer-to-reality geometry type modeling is necessary for liquid passages whenever phase changes occur. Modeling the evaporator may be improved by analyzing with the number of sections rather than one. These are immediate concerns to improve in the modeling significantly. Assumptions made for the system should also be removed one by one. Specifically, assumptions in association with the geometry in three dimensions should be removed as much as possible.

CONCLUDING REMARKS

A numerical simulation model was constructed to investigate the thermal and flow nodes behaviors with respect to energy balance, temperature, and quality distributions of the system under different initial surrounding temperatures. As the system reached the steady state, a unique value of energy for the CPL system indicated that the simulation technique is on the right track. Simulated temperature behaved with a similar tendency to that of the observation with less temperature reduction. The magnitude of the reduction in temperature and quality variations indicate that further refinement solution technique is needed. In the DISCUSSION Section, immediate concerns and necessary steps for further improvement are discussed and identified. A knowledge of quality variation along the passage while phase changes take place is unknown and will be very important for this type of complex thermal fluid problem.

The local acceleration due to the sudden change of cross-sections in a conduit and elevation changes should be studied in detail from the pressure and temperature distributions points of view while phase changes take place.

NOMENCLATURE

ARLXCA	:Allowable arithmetic node relaxation temperature change
ARLXCC	:Computed arithmetic node relaxation temperature change
CAPPMP	:Macro command for evaporation
DRLXCA	:Allowable diffusion node relaxation temperature change
DRLXCC	:Computed diffusion node relaxation temperature change
FASTIC	:Subroutine for a steady state solution
G	:Conductance of conduction or convection of nodes

HX :Macro command for heat exchanger
 LINE :Macro command for tubing
 LOOPCT :Limit of iterations in FASTIC or STDSTL
 OUTCALL :Subroutine for output
 Q_i :Rate of heat of the i-th node
 T_k :Temperature at the k-th iteration
 T_{k+1} :Temperature at the k+1st iteration
 X :Quality defined as the ratio of the vapor mass to the mass of the vapor and liquid

Subscripts

i :Location of nodes
 in :Inlet
 out :Exit

Superscripts

k :Event of iteration in association with LOOPCT
 k+1 :Subsequent event of iteration

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